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(54) ROTARY FLUID-MACHINE

(71) I, GEORGE JAMES DOUNDOULAKIS, a citizen of the United States of America, of 2498 Kayron Lane, N. Bellmore, Long Island 11710, State of New York, United States of America, do hereby declare the invention, for which I pray that a patent may be granted to me, and the method by which it is to be performed, to be particularly described in and by the following statement:-

This invention relates to a rotary fluid-machine such as a pump, a compressor, an air motor, internal and external combustion engines and the like.

The commonest type of cylinder/piston combination for accomplishing compression and/or expansion of fluids in such a rotary fluid-machine is a fixed cylinder containing a reciprocable sliding piston. The working fluid is allowed to enter the cylinder and later is expelled through valves which open and close passages at predetermined intervals, timed with respect to the rotation of a crankshaft. Usually, a plurality of such cylinder/piston combinations constitute such a device.

Axial motion of the pistons of such a device is usually associated with a crankshaft to which the pistons are connected via connecting rods. In the case of pumps and compressors, external rotational torque is applied to the crankshaft, which in turn sets the pistons into axial motion via connecting rods. In the case of steam engines, the axial motion of the pistons is converted into rotation of the crankshaft via connecting rods. In the case of internal combustion engines using axial pistons rotational energy from the fly-wheel of the engine is converted into axial motion of the pistons for compressing a fluid, such as air or a mixture of air and fuel. Subsequently, upon ignition of the fuel, the axial motion of the pistons is converted into rotational torque on the crankshaft. The connecting rods between the pistons and the crankshaft are thus used to transfer energy in both directions, from the crankshaft to the axial pistons and vice versa.

Unfortunately, the conversion from linear to rotational motion is not accomplished without penalty. Thus, for each piston, as the force towards the crankshaft lies along the direction of its connecting rods, while the piston force is directed along the axis of the cylinder, a force component, directed towards the piston rings, exists to balance those two forces. Forces directed towards the side of the cylinder contribute a frictional force (of the order of 14 lbs. per square inch of piston area), opposing the motion of the piston.

As a single cylinder with its piston makes up an engine, most of the present day internal combustion engines, are groups of such engine units. However, cylinders do not fit nicely next to each other. In addition, several other components such as a crankshaft and pairs of popping valves are forced to work together. The well known V-8 arrangement represents an improvement over the more usual vertical cylinder configuration and accurate balancing has eliminated much of the objectionable vibration generated by the linear motion. Nevertheless, multi-cylinder piston engines utilise space very poorly. They are also very heavy.

Over the years, since Watt's time, several attempts have been made to improve efficiency with rotary arrangements where the pressure inside the cylinder acts on radial pistons around the output shafts, so that interaction takes place directly in terms of torque. The turbine is one such attempt. The Wankel engine is another such attempt. Still another known type of rotary engine design is based on what has been referred to as the 'scissor-action' pistons, shown in Figure 1. While various arrangements have been set forth over the years for the control of the 'scissor-action' pistons, the pistons

themselves, as a rule, are supported from around a centre shaft 20 by telescoping shafts such as 18 and 19 in an arrangement resembling that of a two-section door hinge, where the flat sections represent pistons 22*a*, 22*b*, 22*c* and 22*d*. The pistons 22*a*, 22*c* and 22*d* operate inside a stationary angular cylinder 24, which is part of a housing 26. Chambers 28*a*, 28*b*, 28*c* and 28*d* are formed between the housing 26 and the telescopic shafts 18 and 19, the size of these chambers being determined by the angular separation of the pistons 22*a*, 22*b*, 22*c* and 22*d*. When the telescoping shafts 18 and 19 hold a pair of pistons, as is usually the case, the four strokes of the Otto Cycle, namely intake, compression, power, and exhaust can be executed. To accomplish expansion and contraction of the chambers, one pair of pistons 22*a* and 22*c* are held fixed while the second pair of pistons 22*b* and 22*d* execute a 90° rotation in a particular rotational direction. Then the second pair of pistons 22*b* and 22*d* is held fixed while the first pair of pistons 22*a* and 22*c* rotatets 90° in the same rotational direction. This action is commonly referred to as 'cat-and-mouse' action.

'Scissor-action' engines have actually been built and, reportedly, have demonstrated impressive results, mainly in terms of power-to-weight ratio. At the same time 'scissor-action' engines have demonstrated these weaknesses:

(a) The pistons are held by the telescoping shafts 18 and 19 along only half their heights for example, the pistons 22*a* and 22*b* are held along portions 32*a* and 32*b*, respectively, and are cantilevered over the other halves 33*a* and 33*b* of their heights. Thus, along the portions 33*a* and 33*b* of these pistons there is no support. The pistons thus provide inherently weak structures unless they are substantially massive. Increased mass, however, leads to an increase in the moment of the inertia of the pistons and, therefore, a reduction in their capability for fast acceleration.

(b) Sealing between adjacent chambers becomes a problem. This leads to two alternatives:

(i) If no special sealing elements are used, a space must be left between pistons 22*a*, 22*b*, 22*c* and 22*d* and the case of the housing 26 to preclude locking of the pistons and the housing owing to unequal thermal expansion. This space contributes to loss of pressure and fuel.

(ii) If sealing elements such as 42*a* and 42*b* (corresponding to piston rings) are used, difficulties arise in lubricating these elements. In the case of internal combustion engines, if oil is burned with petrol, a continuous supply of oil must be provided, with the chance of increase in pollution products. If the oil is to run around the pistons, between the sealing elements 42*a* and 42*b* and the housing 26, the oil can spill into the intake and exhaust ports, such as port 45, as well as into the spark plug opening, as the radial pistons pass over such ports.

Sealing each piston along 7/8 of its periphery, presents difficulties. Sealing becomes easier in toroidal configuration but the problem of oil spilling in the ports still remains.

It is usually the passage of the pistons over the ports which determines the opening or closing of a port, in the case of 'scissor-action' engines, rather than the use of popping valves. Wide and usually massive pistons are then needed properly to time intake and exhaust, with the sealing problems still outstanding.

A three-rotor engine has been proposed previously, and shows that an improved rotary engine can be accomplished if both pairs of interlevered pistons are in simultaneous motion with respect to the body of the engine. However, this proposed engine suffers from the fact that the internal wall surface of the cylinder in which the radial pistons operate, belongs to three separate members of the device, namely, to the two rotors which are used to hold the interleaved radial pistons, a first rotor providing the inner cylindrical wall surface and one flat end wall, the second rotor providing the other flat end wall of the cylinder, and to the stationary case of the engine which provide the outer cylindrical surface of the cylinder in which the radial pistons operate. The weakness in this design lies in the fact that, during the fuel explosion, the internal surface of the engine would tend to be taken apart by the internal pressure. Thrust bearings would, therefore, have to be used between the side rotors and housing to prevent this happening, but the use of such thrust bearings would lead to considerable friction losses.

Another weakness of the cylinder design in this three-rotor engine, is the excessive moment of inertia of the accelerated rotors. This moment of inertia is contributed by the rotor forming the outer base of the cylinder in which the radial pistons operate.

The present invention provides a rotary fluid-machine comprising a fixed housing, a cylindrical drum within the housing and having an outer cylindrical wall and two end walls, first radial pistons rigidly attached to the inside of the outer cylindrical wall of the drum, each of the first radial pistons separating two mutually adjacent chambers inside the drum from each other, a rotor coaxial with, and internal to, the cylindrical drum, a shaft coaxial with, and internal to said rotor, the shaft being connected to said drum for rotation therewith, second radial pistons rigidly attached to the rotor, equal in number to the first radial pistons and interleaved with

the first radial pistons, each of the second radial pistons separating two mutually adjacent chambers from each other, an annular space between the rotor and the outer cylindrical wall of the drum being divided by the first and second radial pistons into said chambers, of which the total number is equal to the total number of the first and second radial pistons, half of the chambers being, in use, contracting and the other half of the chambers being expanding upon a relative rotation of the drum and rotor, intake and exhaust means through which a fluid can pass into the expanding chambers and from the contracting chambers, and means rotatably supporting the drum and the rotor within the housing.

With the inner rotor and the drum mutually supported on bearings, the side walls of the contracting/expanding chambers, formed inside the drum, can withstand high pressures without the danger of coming apart and without the need for thrust bearings to prevent explosion of cylinder walls in the axial direction.

The first (outer) radial pistons are rigidly attached to three sides inside the drum so that there is sliding contact only between the inner sides of the outer radial pistons and the cylindrical wall surface of the inner rotor.

The (second) inner radial pistons having a relatively low moment of inertia, are connected to the inner rotor along their entire axial length and slide with respect to the inner drum wall on three of their sides.

The drum which provides the cylinder in which the radial pistons operate, is rotatably supported by the outer wall of the housing and rigidly connected to a centre shaft. Expansion and contraction of the chambers is accomplished by the inner rotor which holds the inner radial pistons, accelerating from zero speed with respect to the housing to a speed twice the speed of the uniformly rotating drum which holds the outer radial pistons.

Intake and exhaust can be accomplished by means of ports which are cut in the outer end wall of the drum coming into coincidence with slots azimuthally cut in the stationary end wall of the housing, with which the outer end wall of the drum is in continuous sliding contact. The inner radial pistons do not cross the ports, they simply effectively oscillate between the outer radial pistons which are attached inside the constantly rotating drum. In this rotary fluid-machine, lubricating oil is permitted to circulate between pairs of sealing elements around the sliding edge of the radial pistons, without the danger of spilling into the input or exhaust ports.

It will be apparent that this rotary fluid-machine has many advantages. Thus, it provides an efficient angular cylinder/piston combination for accomplishing compression and/or expansion in such devices as pumps, compressors and external and internal compression engines. It also provides a plurality of expansion/compression chambers inside a single cylinder where only rotational motion takes place, thereby eliminating the need for axial motion and the piston friction forces associated with the conversion between axial motion of the pistons and rotational motion of the crankshaft. Moreover, the radial pistons are rigidly held along their entire axial length, resulting in small, light and strong pistons. Furthermore, as the drum is separate and free from the fixed housing, this rotary fluid-machine can rotate with respect to the housing which results in a more efficient utilisation of space.

Another advantage of this rotary fluid-machine is that the accelerated pistons do not have to cross the intake and exhaust ports or the ports used for introducing a spark, thereby allowing the facility of lubricating oil to run between sealing elements and the internal surface of the drum around the radial pistons, without spilling into the intake or exhaust ports. Moreover, the accelerating pistons have a low moment of inertia which enables changes in the volume of the chambers to occur rapidly.

The invention also provides a three-rotor device incorporating the type of rotary fluid-machine defined above. Such a device can be used as an efficient pump, a compressor and external or internal compression engine.

The invention will now be described in greater detail, by way of example, with reference to Figures 2 to 9 of the accompanying drawings, in which:-

Figure 2 is a perspective view of a rotary fluid-machine constructed in accordance with the invention, a portion of the external housing and the drum wall being shown broken away to reveal the internal parts;

Figure 3 is a part-sectional side elevation of a tri-rotor device comprising two rotary fluid-machines constructed in accordance with the invention;

Figures 4*a* and 4*b* are part sectional plan views taken on line 4-4 of Figure 3, showing two extreme positions of the unit interlinking the two cylinders of the device;

Figure 5 is a plan view of the tri-rotor device of Figure 3 showing rotor motion regulating means;

Figure 6 is a part-sectional plan view taken on line 6-6 of Figure 2 (also corresponding to line 6-6 of Figure 3);

Figure 7 is a part-sectional plan view taken on line 7-7 of Figure 2 (also corresponding to line 7-7 of Figure 3);

Figure 8*a* and 8*b* are cross-sections taken on lines 8*a*-8*a* and 8*b*-8*b* respectively, of Figure 2; and

Figures 9*a* and 9*b* are cross-sections taken on lines 9*a*-9*a* and 9*b*-9*b* of Figure 2.

Referring to Figure 2, a rotary fluid-machine 50 is mounted in a stationary housing 52, the machine comprising a cylindrical drum 60 having an outer cylindrical wall 62 and flat end walls 61 and 63. The machine 50 is provided internally with a pair of outer radial pistons 54*a* and 54*b* which are an integral part of the external cylindrical wall 62 and the bases 61 and 63 of the drum 60. The outer radial pistons 54*a* and 54*b* are thus supported on three sides by the walls 61, 62 and 63 of the drum 60. The drum 60 is rotatably supported by a bearing 64 (not shown in Figure 2 but shown in Figure 3 as 64*a* and 64*b* and in Figure 7 as 64*b*). The drum 60 is rigidly connected to a centre shaft 58. However, it is also possible for the drum 60 to be rotated from the outside, for example, by means of gears (not shown), in which case a gear or equivalent gear teeth would be attached to the drum 60.

Internally, between the end walls 61 and 63 and concentrically with the drum 60, the machine 50 contains an inner rotor 65 having rigidly attached to it inner radial pistons 56*a* and 56*b*. The inner rotor 65 is radially supported on a housing 66 by means of a bearing 72 (not shown in Figure 2, but shown in Figure 3 as bearing 72*b*). The centre shaft 58 is kept radially aligned with respect to the machine 50 by means of bearings 73 and 74 (not shown in Figure 2, but shown in Figure 5 as 73*a*, 73*b*, 74*a*, 74*b*). The axial position of the rotor 65 is held fixed and the rotor is prevented from moving axially by means of steps 70 and 71 which slide on the internal surfaces of the drum end walls 61 and 63, respectively. Such sliding contacts may be direct or through thin washers of special bearing material having better frictional properties than the metals of the drum 60 and the inner rotor 65.

Fluid intake and exhaust ports 75*a*, 75*b* and 76*a*, 76*b* are constituted by apertures in the end wall 63 of the drum 60. Ports 75*a*, 75*b*, 76*a*, 76*b* together with exhaust azimuthal slots 81*a*, 81*b* and 81*c* and 81*d* respectively are shown in Figure 6. Intake slots 80*a*, 80*d* and 80*b* and 80*c* are joined through tunnels 79*a* and 79*b* respectively. Also exhaust slots 81*a*, 81*c* and 81*b*, 81*d* are joined through internal tunnels 79*c* and 79*d* respectively. Groups of slots then communicate with the surface of the housing through internal tunnels in the bases of the stationary housing. For example, the exhaust slots 81*a* and 81*c* and the internal tunnel 79*c* communicate with an aperture 78*b* in the stationary housing.

The outer pistons 54*a* and 54*b* of the drum 60 are interleaved with a pair inner radial pistons 56*a* and 56*b* connected to the inner radial pistons 56*a* and 56*b* connected to the inner rotor 65. It is also possible for more than two radial pistons to be attached to the drum 60, in which case an equal number of radial pistons would be attached to the inner rotor 65. The four radial pistons 54*a*, 54*b*, 56*a* and 56*b* divide the machine 50 into four chambers 90*a*, 90*b*, 90*c* and 90*d*. If during the instant shown in Figure 2 the drum 60 is assumed to be rotating anti-clockwise in the direction of the arrow 91 with an angular velocity Wo, and the inner rotor 65 (with the pistons 56*a* and 56*b*) to be rotating in same rotational direction, shown by an arrow 92, with angular velocity 2Wo, then chambers 90*a* and 90*c* will be contracting, and the chambers 90*b* and 90*d* will be expanding.

Tight pressure separation between the chambers 90*a*, 90*b*, 90*c* and 90*d* is accomplished by means of sealing elements positioned around the edges of the radial pistons 54*a*, 54*b*, 50*a*, 50*b* along the sliding surfaces. The preferred design for such sealing elements is shown in Figures 8*a*, 8*b* and 9*a*, 9*b*. Figures 8*a* and 8*b* show 'U' shaped double-blade sealing elements 92 and 93 for sealing three sides of the inner pistons 56*a* and 56*b*. The blades 92 and 93 each have one leg 97 narrower than the other leg 98, the legs of the sealing elements being such that one narrow blade leg and one wider blade leg are presented at both the top and the bottom of the pistons. Springs 94, 95 and 96 are provided to urge the blade 92 towards one end wall 61 of the drum 60, and to urge the blade 93 towards the other end wall 63 of the drum. Both blades 92 and 93 have the same width along the intermediate sections. A spring 97′ is provided to urge both blades towards the internal surface 98′ of the drum 60. A small space is left between the ends of the legs 97 and 98 of the sealing elements 92 and 93 and the ends of slots 99*a* and 99*b* cut in the inner rotor 65 to take up temperature expansion of the sealing elements.

Figures 9*a* and 9*b* show double-blade sealing elements 102 and 103, which are similar to those shown in Figures 8*a* and 8*b* for sealing the sides of the outer radial pistons 54*a* and 54*b* with respect to the cylindrical surface of rotor 65. These 'U' shaped blades 102 and 103 also have legs of unequal widths and are placed together in such a way that one wide leg rests next to a narrow leg. Springs 104 and 105 are provided to urge the blade 102 towards the end wall 61 of the drum 60 and to urge the blade 103 towards the end wall 63 of the drum 60. A spring 107 is provided to urge both blades 104 and 105 towards the cylindrical surface of the inner rotor 65. Small clearances 109*a* and 109*b* are left at the two extremities of the sealing elements for temperature expansion.

The machine 50 described above can be used in the design of various devices such as compressors, pumps, steam engines, geothermal engines, air motors, hydrostatic pressure engines, internal combustion

engines and the like.

If the working fluid taken into the expanding chambers 90*b* and 90*d* through an intake port comes under pressure higher than the pressure in the exhaust port, force can be exerted on the surface of the radial pistons causing them to be angularly displaced. When a structure is angularly displaced by an angle $d\theta$ under a torque T the work performed by the fluid on the system is $dw = Td\theta$. This work, as it will be explained later in connection with Figure 3, can be provided as an output on the body of the drum 60, and therefore on the output shaft 58, when the drum is directly connected to the shaft 58. In this case, the machine 50 acts as an engine converting external pressure, present in the working fluid, into torque at the output shaft 58. When the fluid is steam under pressure, the device acts as a steam engine. When the fluid is a geothermal gas under pressure the engine acts as a geothermal engine. In such cases, the energy in the steam or geothermal gas is converted by the engine into torque. This engine can also act as an air motor when the pressure comes from a high pressure tank whose energy is being converted into torque.

When the fluid is a liquid under pressure, for example water under a hydrostatic pressure, the engine acts as a high efficiency hydrostatic torque generator. Since rotation of the drum 60 can be made to correspond exactly with the amount of fluid passing through the machine 50, the device can also be used as a fluid measuring unit as it would be a watermeter or a petrol pump.

When the purpose of the machine 50 is to raise the pressure of the working fluid to a higher pressure at the exhaust port, the device will act as a compressor. Such compressors can be used in air-conditioning installations, refrigerators, freezers, dehumidifiers and the like. Compressors are also used for storing high pressure in pressure tanks for driving automatic machinery, for tyre inflation at petrol stations, and the like. It should be noted that the work involved in compressor and pump applications must be externally provided to the machine 50 through the shaft 58, or directly through gears to the drum 60. It may further be noted that the machine 50 may be used to provide a vacuum pump by connecting the intake port to a vacuum tank.

The rotary fluid-machine 50 may also be used as part of an improved tri-rotor device. Figure 3 shows an improved tri-rotor device including two rotary fluid-machines 50*a* and 50*b* mounted inside a stationary housing 52. An interlinkage unit 110 (shown in detail in Figures 4*a* and 4*b*) serves to interlink the two machines 50*a* and 50*b*. Rotor control mechanisms 112*a* and 112*b* (shown in greater detail in Figure 5) serve to regulate the motion of the inner rotors 65*a* and 65*b* of the two machines 50*a* and 50*b* as a function of the angular rotation of the shaft 58. Both the drums 60*a* and 60*b* of the two machines 50*a* and 50*b* are rigidly connected to each other by spacing posts 113 and 114, and by a cover cylinder 115. The spacing posts 113 and 114 are rigidly connected to a plate 116, which in turn is rigidly connected to the centre shaft 58. Therefore, the two drums 60*a* and 60*b*, the cover cylinder 115, the spacing posts 113 and 114, and the shaft 58 all rotate together, preferably at a substantially uniform speed, about the axis 1-1 of the device 109.

The centre shaft 58 is rotatably supported on the two inner rotors 65*a* and 65*b* by means of pairs of radial bearings 73*a*, 74*a* and 73*b*, 74*b* respectively. The drums 60*a* and 60*b*, which are integrally formed with the centre shaft 58, are also rotatably supported on the inner rotors 65*a* and 65*b* by means of radial bearings 64*a* and 64*b*, respectively. The entire system just described, comprising the centre shaft 58, the two rotors 65*a* and 65*b*, the two drums 60*a* and 60*b*, the cover cylinder 115 and the spacer posts 113 and 114, is then rotatably supported in the stationary housing 52 by means of two bearings 72*a* and 72*b*.

The interlinkage unit 110 acts as a differential unit interconnecting the inner rotor 65*a* with the inner rotor 65*b*. Alternatively, the unit 110 prevents the rotor 65*a* from rotating while the rotor 65*b* is free to rotate, then the rotor 65*b* is prevented from rotating while the rotor 65*a* is free to rotate, each of these phases occurring during the time interval that it takes for the constantly rotating drums 60*a* and 60*b* to rotate substantially through 90°.

Assuming that the tri-rotor device 109 acts as a steam engine at the particular interval when the rotor 65*a* is free to rotate while the rotor 65*b* is held fixed. In the drum 60*a*, each inner radial piston will receive a torque F so that the rotor 65*a* will receive a total torque of 2F and since the rotor 65*b* is held fixed, the output shaft 58 will effectively see a torque of 4F in the positive rotational direction. Simultaneously, each outer radial piston will see a torque of -F since it follows the pressure-providing steam. A total of -2F is, therefore, received by the two outer pistons. Since the outer pistons are directly connected to the shaft 58 the torque of -2F from the outer pistons will be equally reflected to the output shaft 58 as -2F. Therefore, the total output torque of the drum 60*a* will be 4F-2F = +2F. The rotary fluid-machine 50*b* will also provide a torque of +2F, F on each outer radial piston, which in this case precedes the pressurised steam. The inner radial pistons of rotor 65*b* will provide zero torque during this interval as they are held fixed. During the next interval, the function of the rotary fluid-machines 50*a* and 50*b* will be interchanged,

with rotor 65*a* being held fixed. But, regardless of the time interval being considered, each machine 50*a* and 50*b* will contribute a torque of 2F to the output shaft 58. Therefore, the output shaft 58 will receive a total torque of 4F from the two drums 60*a* and 60*b*. In general, the output torque is equal to the torque applied to the surface of a piston multiplied by a half of the total number of the pistons.

Figure 3 shows that the rotors 65*a* and 65*b* terminate towards the centre of the devie 109, into plates 120*a* and 120*b*, respectively. The operation of the interlinkage unit 110 is best shown in Figures 4*a* and 4*b*. The plates 120*a* and 120*b* are connected, via connecting rods 122*a* and 122*b* to a rocker 121 which is pivoted about a shaft 123. The shaft 123 may be one of the spacer posts, such as 114 rigidly supported by both drums 60*a* and 60*b*. When the rotor 65*a*, together with the plate 120*a*, rotates anti-clockwise, the connecting rod 120*a* pulls the rocker 123, forcing it also to rotate anti-clockwise at substantially the same angle. Simultaneously, the connecting rod 122*b* urges the plate 120*b* in the clockwise direction. Therefore, the rotation of the rocker 128 in a particular direction urges the two inner rotors 65*a* and 65*b* to rotate in opposite directions. However, if one of the inner rotors is held fixed, the motion of the other inner rotor causes the assembly of the drums 60*a* and 60*b* and the shaft 58 to rotate in the same direction as the free inner rotor. After the rotors reach the extreme position shown in Figure 4*a*, as the rocker 121 rotates in the direction of the arrow 124*a* the fixed rotor is released, and the rotor which was free now becomes fixed. The rocker 121 now reverses its direction of motion, going in the direction shown by the arrow 124*b* in Figure 4*b*. The final relative position of the rotor plates 120*a* and 120*b* and the rocker 121 after the rocker rotates through 90° is shown in Figure 4*b*. It should be noted that, since only one of the rotor plates 120*a*, 120*b* moves during each displacement, the relative positions of the rotor plates shown in Figures 4*a* and 4*b* are attained because of the actual rotation of the drums 60*a* and 60*b* which carry the rocker 121, in the anti-clockwise direction.

The means used to hold one inner rotor fixed while allowing the other rotor to move forward can be a very involved mechanism. The sophistication involved in the design of such a mechanism can greatly contribute to the overall efficiency of the device.

Figure 5 shows simple basic means for stopping one inner rotor while allowing the other to move forward. Figure 5 shows the rotor 65*a* provided with ratchet steps 128*a*, 128*b*, 129*a* and 129*b*. The pair of steps 129*a* and 129*b* interact with forward pawls 131*a* and 131*b* to prevent the inner rotor 65*a* from rotating anti-clockwise during the time it takes for the centre shaft 58 to rotate 90°. The anti-clockwise direction of rotation is assumed to be the positive direction of rotation. Assuming steam enters the two chambers of the drum 60*a* during the interval in which rotor 65*a* is held fixed, the pressure of the steam will urge the inner radial pistons to reverse their motion and move in the negative direction. The ratchet steps 128*a* and 128*b* of the inner rotor 65*a* will then interact with the pair of rear pawls 132*a* and 132*b* and prevent the rotor 65*a* from moving in the negative direction. A cam 130, rigidly attached to, and rotating with, the shaft 58 is shown in Figure 5 at the instant when it acts upon rollers 135*a* and 135*b* rotatably supported on the pawls 131*a* and 131*b* by means of shafts 134*a* and 134*b* for displacing the pawls radially outwardly and thereby disengaging the inner rotor 65*a* from the forward pawls 131*a* and 131*b*. The inner rotor 65*a* will then rotate through 180° before it becomes re-engaged with the pawls 131*a* and 131*b*, while the centre shaft 58 will be displaced through 90°. During the next 90° rotation of the centre shaft 58, the inner rotor 65*a* will remain engaged. It will again be disengaged at the end of such 90° rotation of the centre shaft 58. This means that the inner rotor 65*a* remains engaged with the pawls 131*a* and 131*b* during every other 90° rotation of the centre shaft 58. A similar mechanism (not shown) acting on the inner rotor 65*b*, keeps this rotor engaged, and therefore fixed, during the 90° intervals during which the rotor 65*a* is free to rotate and vice versa.

Spring action is provided to the system by supporting the pawls 131*a*, 131*b*, 132*a* and 132*b* on a circular ring 140 which is suspended by blade springs 133*a*, 133*b*, 133*c* and 133*d* from a cylindrical ridge 141, which is rigidly connected to the housing 66.

It should be noted that, while the rotation of the centre rotor 58 provides the criterion for engaging and disengaging the inner rotors at the end of each 90° interval, the exact time of release or engagement can be advanced as a function of speed or internal pressure so smooth engagement and disengagement results.

During normal operation, for example, a forward moving rotor, say rotor 65*a*, does not have to be stopped by the forward pawls 131*a* and 131*b*. As the drum 60*a* rotates with an angular velocity Wo, and the forward moving inner rotor 65*a* rotates with an angular velocity 2Wo, the inner pistons are approaching the outer pistons at a relative angular velocity of Wo. If the angular slots 81*a* and 81*b* which provide communication between the contracting chambers and the external sink, end beyond a predetermined angle, as the two radial pistons are approaching each other, the remaining steam

in the contracting chambers will be compressed and will act as a cushion on which the velocity of the inner piston, with respect to the outer piston will be reflected. The +Wo velocity of the inner piston with respect to the outer piston will therefore be reflected as -Wo velocity with respect to the outer piston and with the latter rotating at a substantially uniform velocity Wo with respect to the housing the reflected velocity of the inner rotor will be zero with respect to the housing. At this instant the entire angular momentum of the inner rotor *65a* will have been transferred to the centre shaft 58 which is connected to the load. Assuming that the position of the inner rotor 65 at this time is such that the steps, *128a* and *128b* have just passed the tips of the rear pawls *132a* and *132b*, new steam is introduced, the ports of the expanding chambers will now overlap the intake slots *80a* and *80b*, the rotor *65a* will reverse velocity and travel for a short interval clockwise until it is stopped by the rear pawls *132a* and *132b* as they engage with the inner rotor ratchet steps *128a* and *128b*. It should be noted that the engagement between the ratchet steps *128a* and *128b* and the pawls *132a* and *132b* is very smooth because of the low relative velocity between the inner rotor *65a* and the external housing. Moreover, the plate 140 which holds the pawls *131a*, *131b*, *132a* and *132b* provides spring action as it is suspended by the spring blades *133a*, *133b*, *133c* and *133d*. As the force provided by the steam will store some energy as potential energy in the spring blades, substantially all of such energy will be returned to the system as the pressure of the steam will be lowered owing to the expansion of the chambers. This energy can be used to provide initial motion of the inner piston during its next excursion. Most of the energy required to accelerate the inner pistons will be provided by the rotational momentum of the drum *60a* or *60b*. Because of the interlinkage unit 110, the size of the contracting chambers in the drum *60a* is substantially the same at any instant as the size of the contracting chambers in the drum *60b*. Therefore, as the velocity of a rotating inner piston is reflected from +Wo to -Wo, with respect to the rotating drums, the velocity -Wo of the stationary inner rotor with respect to its drum is simultaneously similarly reflected to become +Wo, corresponding to +2Wo with respect to the stationary housing. This implies that the inner rotor, being slowed down, is being decelerated at substantially the same rate as the other rotor is being accelerated. As the decelerating rotor passes its angular momentum to the rotating drum *60a* or *60b*, the accelerating rotor takes angular momentum at substantially the same rate from the rotating drum. Angular momentum is, therefore, shifted between the rotors *65a* and *65b* while the angular momentum of the drums and shaft assembly *60a*, *60b* and 58 remains substantially uniform. The steam will provide torque to the output shaft 58 through the outer piston and through the interlinkage unit 110.

It is to be noted that instead of the pawls and ratches other equivalent mechanisms such as over-riding clutches, magnetic clutches and the like may be used between each inner rotor and the stationary housing to control the motion of the inner rotors.

In the case where the tri-rotor device 109 is used as a compressor the centre shaft 58 is driven by an external torque, while the pawls serve to regulate the motion of the inner rotors.

WHAT I CLAIM IS:-

1. A rotary fluid-machine comprising a fixed housing, a cylindrical drum within the housing and having an outer cylindrical wall and two end walls, first radial pistons rigidly attached to the inside of the outer cylindrical wall of the drum, each of the first radial pistons separating two mutually adjacent chambers inside the drum from each other, a rotor coaxial with, and internal to, the cylindrical drum, a shaft coaxial with, and internal to said rotor, the shaft being connected to said drum for rotation therewith, second radial pistons rigidly attached to the rotor, equal in number to the first radial pistons and interleaved with the first radial pistons, each of the second radial pistons separating two mutually adjacent chambers from each other, an annular space between the rotor and the outer cylindrical wall of the drum being divided by the first and second radial pistons into said chambers, of which the total number is equal to the total number of the first and second radial pistons, half of the chambers being, in use, contracting and the other half of the chambers being expanding upon a relative rotation of the drum and the rotor, intake and exhaust means through which a fluid can pass into the expanding chambers and from the contracting chambers, and means rotatably supporting the drum and the rotor within the housing.

2. A rotary fluid machine as claimed in Claim 1, wherein the drum is rotated, in use at a substantially uniform speed 'Wo' with respect to the housing, while the inner rotor accelerates and decelerates from zero speed with respect to the housing to a speed '2Wo' with respect to the housing.

3. A rotary fluid-machine as claimed in Claim 1 or Claim 2, wherein the intake and exhaust means comprise slots which are angularly disposed on an end wall of the housing and apertures in the adjacent end wall of the drum, the slots and apertures coinciding as the drum is rotated with respect to the housing.

4. A rotary fluid-machine as claimed in

any one of Claims 1 to 3, where there are two first radial pistons and two second radial pistons.

5. A rotary fluid-machine substantially as hereinbefore described with reference to, and as illustrated by, Figure 2 of the accompanying drawings.

6. A tri-rotor device comprising two rotary fluid-machines mounted in a common housing, each of the angular cylinders being as claimed in any one of Claims 1 to 5.

7. A device as claimed in Claim 6, wherein differential means are provided for interlinking inner rotors of the two rotary fluid-machines.

8. A device as claimed in Claim 6 or Claim 7, wherein the drums of the two rotary fluid-machines are rigidly connected together for substantially uniform rotation with respect to the housing.

9. A device as claimed in Claim 8, wherein the two drums are rigidly connected to a single, common coaxial shaft.

10. A device as claimed in Claim 8 or Claim 9 when appendant to Claim 7, wherein the differential means comprises a rocker, rotatably supported by the two drums, at least one connecting rod for connecting the inner rotor of a first of the rotary fluid-machines to the rocker, and at least one other connecting rod for connecting the inner rotor of the second rotary fluid-machine to the rocker.

11. A device as claimed in any one of Claims 6 to 10, further comprising inner rotor motion regulating means for controlling the motion of each of the inner rotors.

12. A tri-rotor device substantially as hereinbefore described with reference to, and as illustrated by, Figures 3 to 9 of the accompanying drawings.

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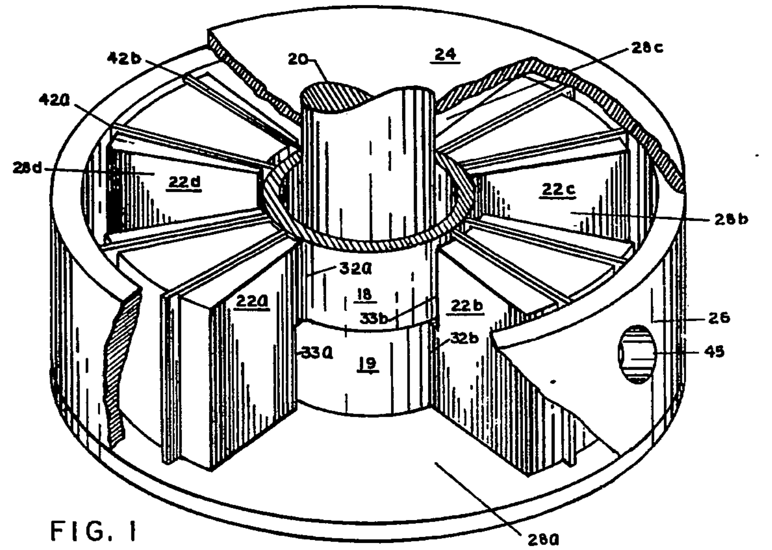
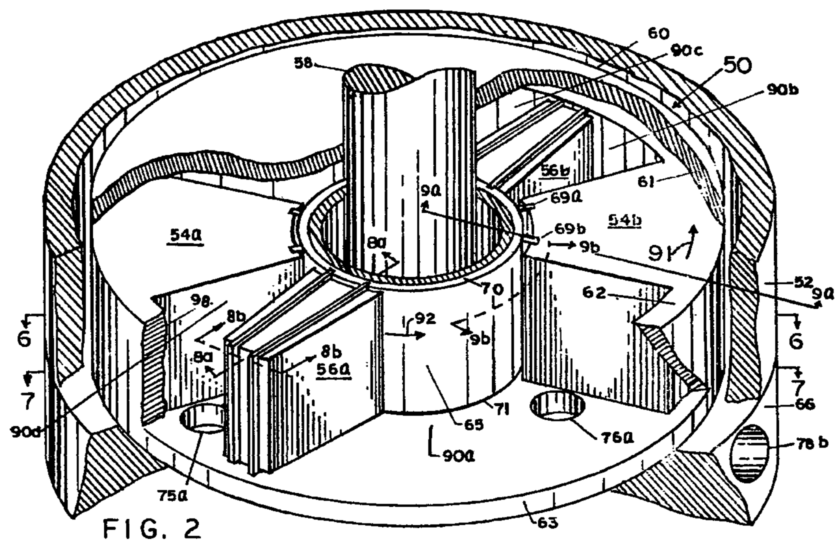

42b
20
24
28c
42a
28d
22d
22c
28b
32a
18
22a
22b
33b
26
32b
33a
19
45
28a

FIG. 1


60
58
90c
50
90b
56b
61
69a
9a
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54a
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9b
9
70
52
9a
62
98
8b
92
6
9b
6
8a
8b
56a
7
7
66
90d
65
71
75a
76a
78b
90a
75a
63

FIG. 2

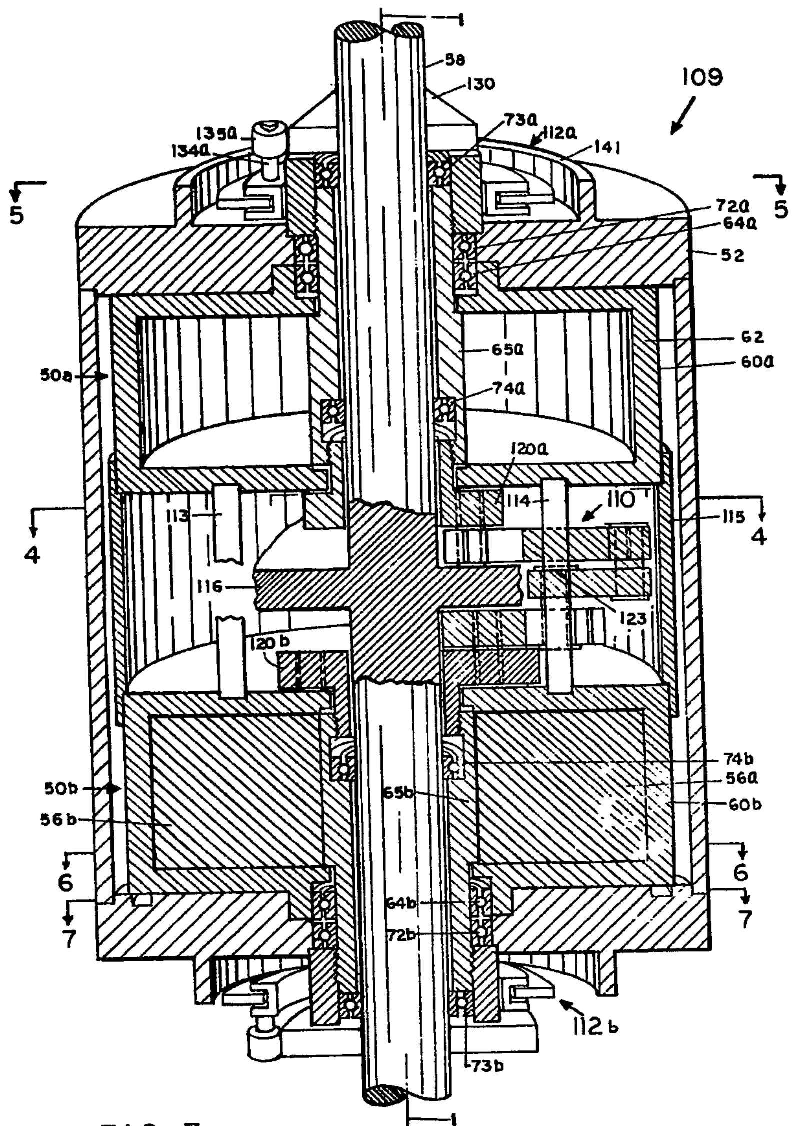
58
130
109
135a
134a
73a
112a
141
5
5
72a
64a
52
62
65a
60a
50a
74a
120a
114
110
113
115
4
4
116
123
120b
74b
56a
50b
65b
60b
56b
6
6
64b
7
7
72b
112b
73b

FIG. 3

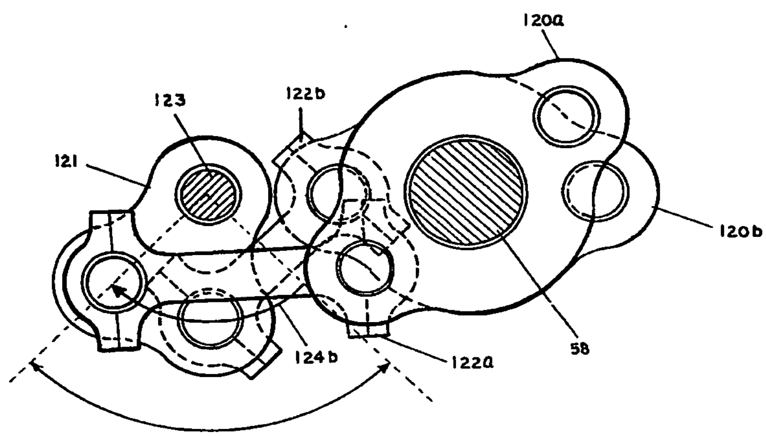
120a
123
122b
121
120b
124b
122a
58

FIG. 4a

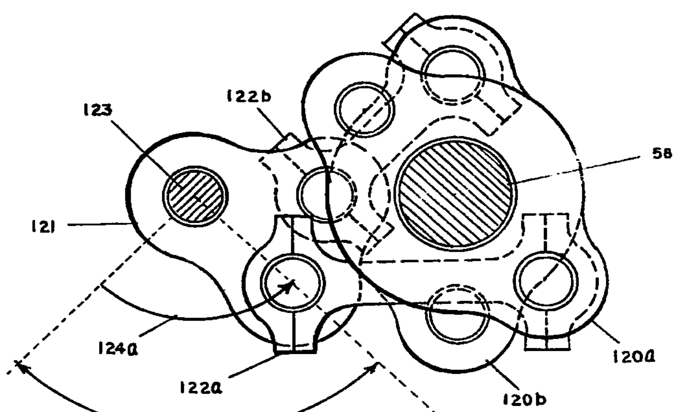
122b
123
58
121
124a
122a
120a
120b

FIG. 4b

1554358 COMPLETE SPECIFICATION

5 SHEETS *This drawing is a reproduction of the Original on a reduced scale*

Sheet 4

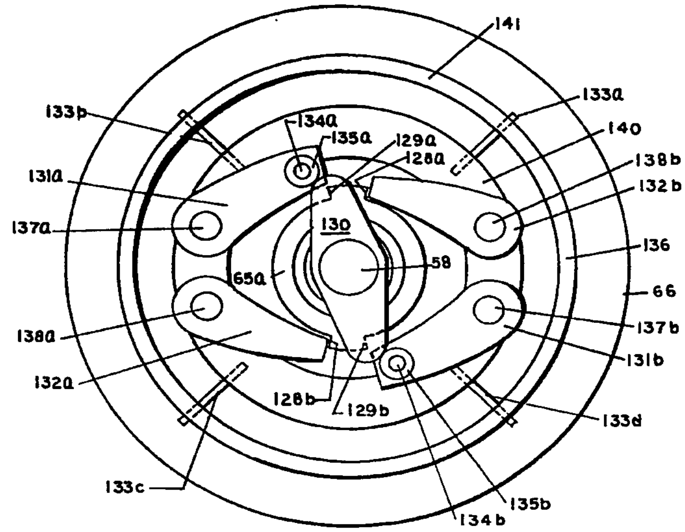

FIG. 5

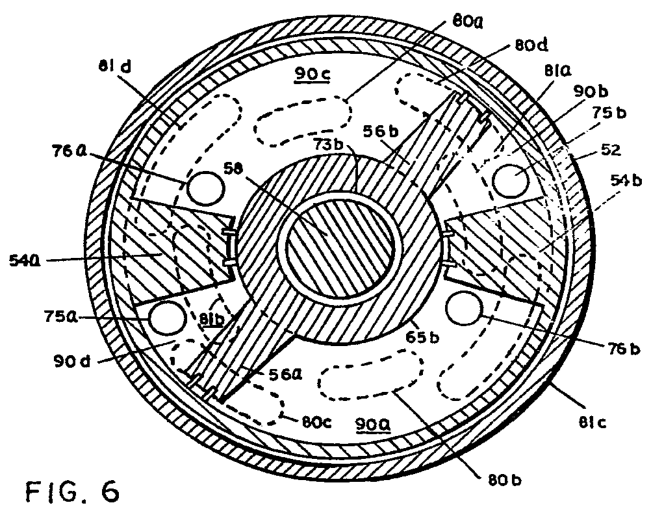

FIG. 6

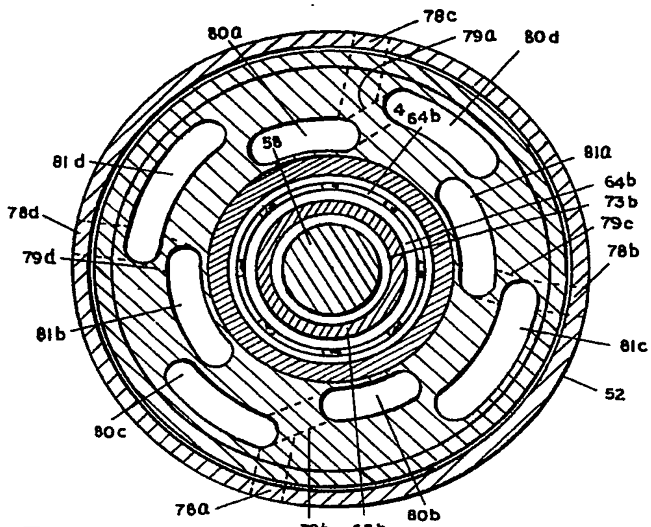

FIG. 7

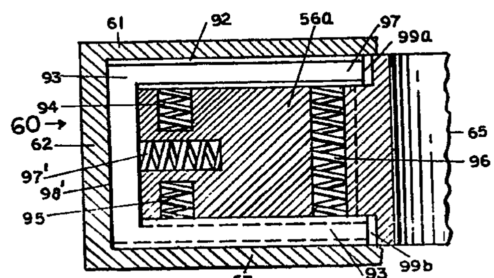

FIG. 8a

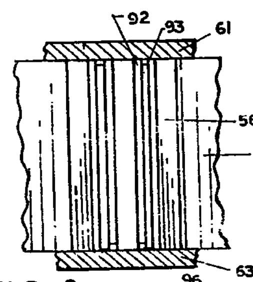

FIG. 8b

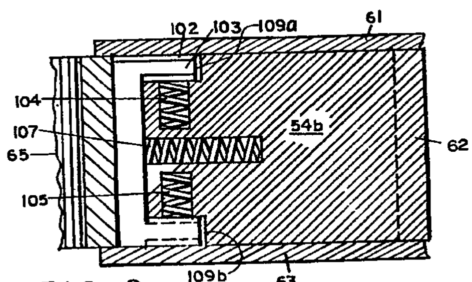

FIG. 9a

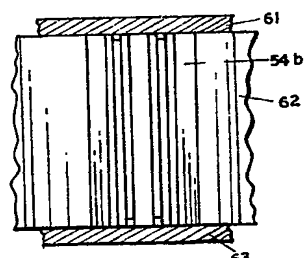

FIG. 9b

Aktenanlage

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Korrespondenz	
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